Coulomb Friction Model Analysis, Acting on an Aerospace Servocommand

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Abstract. Modern flight control system design necessitates using highly detailed models to analyze individual components or subsystems; on the other hand, more fundamental and synthetic models with adequate accuracy are required for preliminary design, monitoring or diagnostic issues.

One of key problems for especially primary flight commands which are depicted as position servo command with high degree of accuracy is the necessity to acknowledge their behavior, under the influence of the dry friction or the friction forces (or their momentum) acting on a generic movable mechanical part of a servo command, or on an entire transmission.

In general this forces depends from the relative velocity of the transmission, or from the exchange load between parts of the mechanism. For this reasons it become very important to take into account these forces or their momentums in relation of their behaviour during the construction of the detailed mathematical models.

In this work we present some generic considerations of the dry friction and dynamic friction applied on an aerospace servo command, with the related mathematical model run in MATLAB/Simulink, and compared to previously tests done for this servo command.

Keywords: Coulomb friction, aerospace servo command, mathematic model, MATLAB/Simulink.

1 Introduction

Modern flight control system design necessitates using highly detailed models to analyze individual components or subsystems; however, more fundamental and synthetic models with adequate accuracy are required for preliminary design, monitoring, or diagnostic issues [20]. One of the key problems for especially primary flight commands, which are depicted as position servo command with a high degree of accuracy, is the necessity to acknowledge their behavior under the influence of the dry friction, i.e., the friction forces (or their momentum) act on a generic movable mechanical part of a servo command or an entire transmission. In general, these forces depend on the relative velocity of the transmission or the exchange load between parts of the mechanism. For this reason, it becomes essential to consider these forces or their momentums about their behavior during the construction of a detailed mathematical models [1].

1.1 Dependence of the friction on speed

The marked non-linearity of the relationships linking the trend of the dry friction forces (or their momentum) to the speed (see Figure 1 in this regard) [8], combined with the extreme importance that this phenomenon assumes in many applications, in the course of decades has justified various studies aimed at defining exhaustive mathematical models.

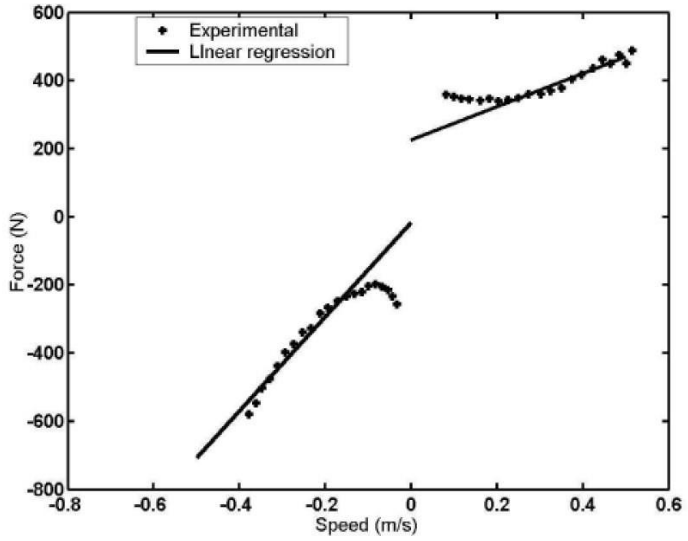

Fig. 1. Trend (experimental) of the friction force acting between surfaces in relative motion; in this case, the measured force is equal to the sum of the dry and viscous terms [6], [8].

Adopting a representation of dry friction to be included in numerical models representative of the dynamic performance in the time domain of mechanical systems, rather than orienting towards mathematical models of tribological derivation [2], [3], [4], it was preferred to opt for a performance model which, ignoring the effects and local dynamics, provides a global representation of the phenomenon; in particular, the classical Coulomb friction model was adopted. Assuming as conventions the positive direction of the friction forces opposite to the positive direction of the speed, the model can be summarized as follows:

$$FF = \begin{cases} F_{att} & if\ v = 0 \wedge |F_{att}| \le FSJ \\ FSJ \cdot sgn(F_{att}) & if\ v = 0 \wedge |F_{att}| > FSJ \\ FDJ \cdot sgn(v) & if\ v \neq 0 \end{cases} \tag{1}$$

The mathematical model of dry friction (1) developed by Coulomb [8], capable of approximating the real physical behavior in a satisfactory way (illustrated in Figure 1), is schematically represented in Figure 2; FF indicates the generic friction force (i.e. its instantaneous value), FSJ the value of the friction force in static or adherent conditions and FDJ represents the friction force in dynamic conditions. This model discriminates the sign of the friction force as a function of the direction of the speed, and is able to distinguish the adherence conditions from the dynamic ones, to evaluate the possible stop of the mechanical element initially in motion, to keep the mechanical element in adherence conditions and, lastly, to evaluate the possible restart of the initially stationary mechanical element.

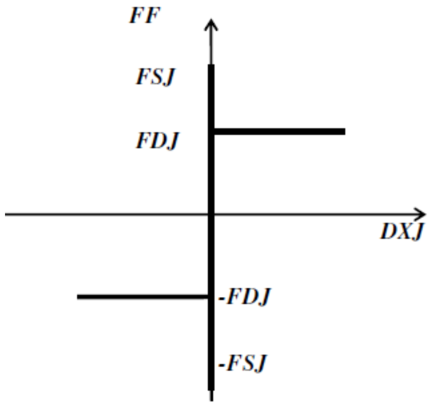

Fig. 2. Friction model schematic representation used in dynamic simulations

At a macroscopic level, the effect produced by dry friction on a mobile mechanical element manifests itself through the creation of forces (or moments) opposite to the relative motion (between the surfaces in contact) and with variable intensity according to the load acting on the element and of the speed itself. In the classical formulation of Coulomb friction, however, it is common practice to model the friction force as dependent only on the speed according to the following model:

- At non-zero speed the force module is assumed constant and equal to the so-called dynamic friction value (FDJ);

- At zero speed the force can assume any value lower than or equal in module to the so-called static friction value (FSJ).

1.2 Dependence of friction on load

As regards the dependence of dry friction on the load acting on the system itself (i.e. the resistant force or momentum applied from the outside to the relative motion of the generic mechanical system), it was necessary to conceive a special model illustrated here after.
At each speed value, the minimum amount of friction force (or momentum) occurs at zero load and its non-zero value is attributable to various causes, in particular to the necessary constructive grip, suitable for containment/recovery/elimination of the mechanical clearance, the use of sealing gaskets in the hydraulic components, and any contacts in electric machines, etc. Furthermore, the presence of loads exchanged between the mechanical connections in relative motion or acting on the transmission increases the minimum value of the friction force (or momentum) by a quantity proportional to the load itself, according to a proportionality coefficient linked appropriately to the efficiency of the connection (or transmission) represented into two different values. Depending on the way of acting, if the motor drags the user (motion against load or "opposing mode", i.e. speed of actuation contrary to the external load acting on the user) or if the user drags the motor (motion in favor of the load or "aiding", i.e. actuation speed in agreement with the external load on the user).
The proposed friction model takes into account, in a synthetic way and in a single formulation, all of the above analyses, moreover, by attributing positive (or negative) values to the efficiency in "aiding" conditions, it simulates reversible (or irreversible) transmissions, also evaluating their level on the basis of the value for the negative efficiency which defines by what percentage the friction force (or momentum) exceeds the load that produces it. Furthermore, the proposed procedure is also capable of taking into account situations such as to produce irreversibility of the mechanical system in static conditions and reversibility in dynamic conditions, simulating its behavior according to reasonable expectations.

2 Consideration for the mathematical model used

The mathematical modeling of the Columbian friction, which is particularly important in the dynamic simulation of mechanical systems (with special regard to position servomechanisms), must therefore be able to describe the behavior of the mechanical element subject to this friction quantified as function of the loads, discriminating between the four possible kinematic situations listed below:

- Mechanical element initially stationary which must remain stationary;
- Mechanical element initially stationary which must start moving;
- Mechanical element initially in motion which must remain so;
- Mechanical element initially in motion which must stop.

The adoption of a model capable of correctly describing these dynamics, considerably increases the accuracy and confidence with the results of the simulations carried out, providing a valid means for investigating critical operating conditions.
The nature of the Coulomb friction cannot be fully described by linear models, and is desirable the possibility of closed-form solutions of the dynamic equations; therefore, any not simplistic modeling must necessarily involve the use of non-linearityes of such complexity as to foreseen the use of numerical resolution techniques based on dynamic simulation in the time domain [9], [11].
The numerical simulation techniques mostly used in the literature, can be traced back to mathematical models which, although more powerful than any purely linear representations, are nonetheless affected by some shortcomings. In particular, the numerical simulation models commonly reported in the literature, albeit with different degrees of approximation, aim to represent this phenomenon by introducing appropriate simplifying hypotheses (different from model to model) aimed at overcoming computational problems related to strict compliance with the Coulomb model. However, this simplifying hypotheses compromise in a more or less important ways the Coulomb model itself [6], [7], [8].

2.1 Proposed Coulomb friction model in relation to the velocity and to the applied active force

The two models proposed, provide a realistic description of the phenomenon and can be easily integrated into more complex calculation programs (related, for example, to servomechanisms or entire actuation systems), also presenting a proven robustness against the particular simulated conditions.
The two numerical simulation models proposed overcome the limits highlighted above and reflect the behavior of the physical model assumed as the objective of the present work; Indeed:

• discriminate the sign of the friction momentum as a function of the direction of the speed,

• distinguish the grip conditions from the dynamic ones (in fact, two distinct values can be assigned for the force – or momentum – of friction, FSJ in static or grip conditions and FDJ in dynamic conditions),

• evaluate the possible stop of the mechanical element initially in motion,

• keep the mechanical element correctly stationary (or in motion) in adherence conditions (or motion),

• evaluate the possible restart of the initially stopped mechanical element,

• moreover, take into account, in a single model, the presence of any limit switches, against which it is assumed that a completely inelastic impact will occur.

The first proposed numerical model, it's referred directly to the Coulomb formulation. It describes the effects of dry friction, as a function of the speed v and the active force F_{att}, according to the modeling represented in Figure 3:

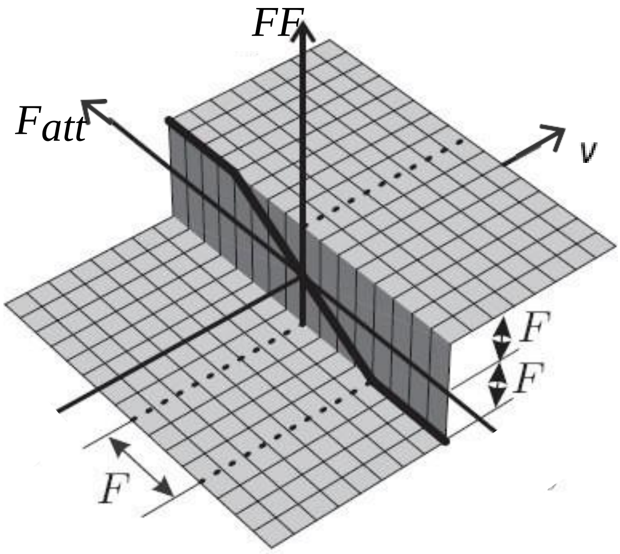

Fig. 3. Graphical representation of the proposed Coulomb friction model

The corresponding mathematical model can be formulated as per equation 1, where:

Where *FF* represents the friction force actually calculated, *FSJ* the value of the friction force in static or adherent conditions, *FDJ* that of the friction force in dynamic conditions and F_{att} the active force applied to the system.

The numerical simulation programs created on the basis of the mathematical model (1), in order not to run into numerical instability phenomena analogous to those already highlighted, are created in such a way as to impose the system shutdown of the mechanical system in the case of speed reversal during the integration steps;

$$v(t_{i+1}) = 0 \quad if \quad v(t_{i+1}) \cdot v(t_i) \leq 0 \tag{2}$$

If this stop results not corrected due to the excess of the active forces F_{att} compared to the passive ones (friction), the next calculation step would generate an imbalance (between the overall forces acting on the system) capable of correctly causing it to restart. It can be stated that the condition expressed in equation (2) constitutes one of the fundamental differences and innovations of the present model with respect to those reported in the literature and, in its apparent obviousness, lies the very robustness and accuracy of the method.

The mathematical model illustrated in equations (1) and (2) is implemented in the MATLAB - Simulink environment in order to obtain a simulation program capable of combining a satisfactory level of confidence and accuracy of the results with high legibility and accessibility.

The transition to the simulation environment supported by MATLAB–Simulink allows to overcome all these problems by providing a symbolic representation of the calculation algorithms adopted which, having a structure formally analogous to the corresponding block diagram, greatly facilitates the understanding of the structure of the dynamic system and, therefore, the legibility of the model itself. In fact, the transition from the aforementioned low-level programming languages to Simulink makes the proposed model accessible to a greater number of users (it is sufficient that the reader has framed the physical problem and is able to read a block diagram). However, this increased accessibility in the other sense introduces loss of the characteristic sequential structure of the algorithm (typical, for example, of the well-known Fortran) and with a significant reduction in the level of readability of the individual calculation operations performed.

The sequential structure, allow to know exactly the trend of the calculation flow (and therefore the order in which, in the single integration cycle, the instructions are executed), considerably facilitates the translation of the mathematical model into the numerical equivalent and allows to solve, in an extremely concise way.

From equation (1) has been developed the subsystem shown below in the figure:

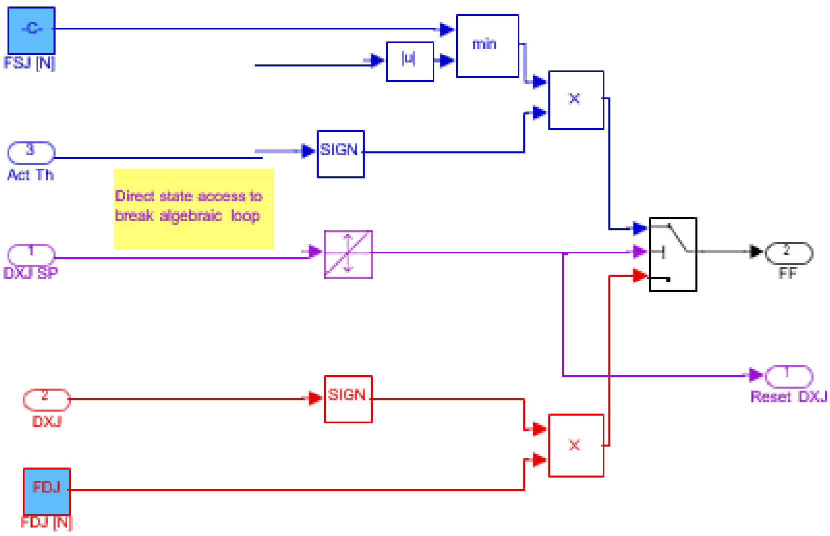

Fig. 4. MATLAB-Simulink block diagram of the proposed friction model

Starting from the subsystem illustrated in figure 4, was developed the non-linear numerical simulation model shown in figure 5:

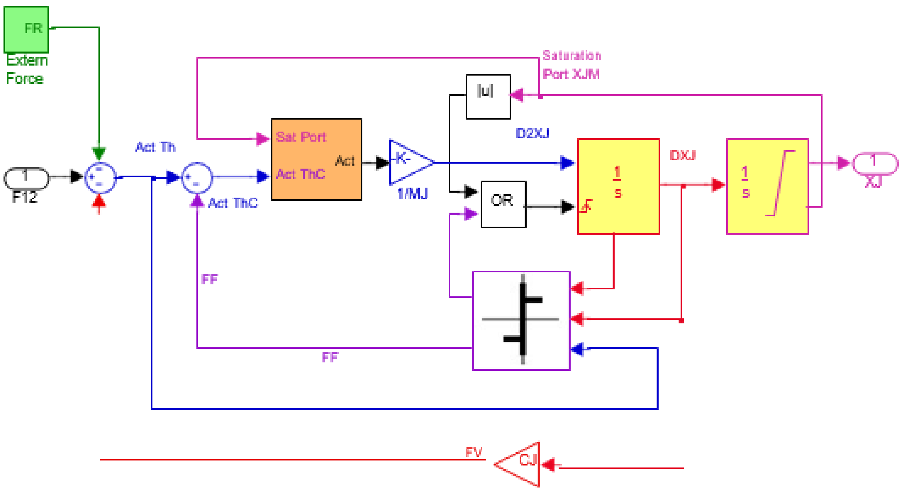

Fig. 5. MATLAB-Simulink block diagram of the actuator model with dry friction and mechanical limit switches

The MATLAB-Simulink block diagram in figure 5, models a typical hydraulic actuator (linear or rotary one) for aeronautical use [13]; this model, based on the active forces F12, the external loads FR and the consequent friction force FF, is capable of simulating the dynamics of the actuator also taking into account any mechanical limit switches [14], [15].

2.2 Coulomb friction model load-sensitive (FR)

The Coulomb friction models available in the literature are usually characterized by extremely simplified structures and limited performances [6], [7], [10]; their limits, easily verifiable through appropriate numerical simulations, are further emphasized by analyzing "integrated" dynamic systems which not only take into account friction, but are also able to simulate the effects generated by the mechanical limit switches and their possible interactions.

The algorithm described in the previous paragraph 2.1, developed in MATLAB – Simulink environment, provides a valid answer to these problems and, through a self-contained subsystem, can describe both the effects produced by friction forces on the dynamic response of a generic mechanical system as their interactions with any mechanical limit switches, in many different applications.

A second numerical model was also developed, sensitive to the aerodynamic load FR and applicable to both reversible and irreversible cases, which allows to estimate more accurately the Coulomb frictions and their effect on the dynamics of aeronautical servomechanisms (in particular flap actuation and control systems) [16].

From the previously illustrated Coulomb friction model, a new numerical simulation model was therefore developed which considers the friction forces (or momentum) as a function of the load FR on the user and in which the friction momentum is seen as the sum of two distinct terms; this sum, depending on the dynamic conditions, can assume two distinct values (static or dynamic):

• A term independent from the external load FR,
• A term proportional to the load through defined efficiency values.

Efficiency is typically defined as the ratio between the output and input powers of a mechanical system; if this is characterized by a constant transmission ratio (very common occurrence in secondary flight controls), the efficiency can be understood as the ratio between the output and input momentum (converted on the same shaft). The efficiency of the system is therefore dependent on the motion conditions (static or dynamic) and on the external load (in favor or against motion). By suitably choosing the value of these returns, it is possible to simulate the performance of reversible and irreversible systems.

The model calculates the friction momentum as follows:

$$T_{FR} = T_{FR0} + \left(\frac{1}{\eta_{opp}} - 1\right) \cdot T_{LD} \tag{3}$$

Under opposing conditions of load to the motion.

$$T_{FR} = T_{FR0} + (1 - \eta_{aid}) \cdot T_{LD} \tag{4}$$

Under load conditions in favor of motion.

When the actuation speed is non-zero and the load T_{LD} opposes the motion, the friction momentum T_{FR} is obtained through equation (3), while when the load is in favor of the motion, the friction momentum T_{FR} is obtained through equation (4); in both cases the momentum T_{FR} assumes the same sign as the actuation speed (seen by the sign conventions) and is equal to the friction momentum $T_{FR,din}$ measured in dynamic conditions. On the other hand, when the actuation speed is zero, the friction momentum T_{FR} is equal to the driving momentum A_{ct} if this is included within two limit values: when the load T_{LD} is positive, the lower (negative) value is deducted from (4), while the upper (positive) one is obtained from (3); if the T_{LD} load is negative, the lower (negative) value is deduced from (3), while the upper (positive) one is obtained from (4); in both cases the term T_{FR} is equal to the friction momentum $T_{FR,stat}$.

This mathematical model was then implemented in the MATLAB - Simulink environment and integrated within a more complex calculation algorithm, realizing the numerical simulation program illustrated in the figure 6, [17], [18], [19]:

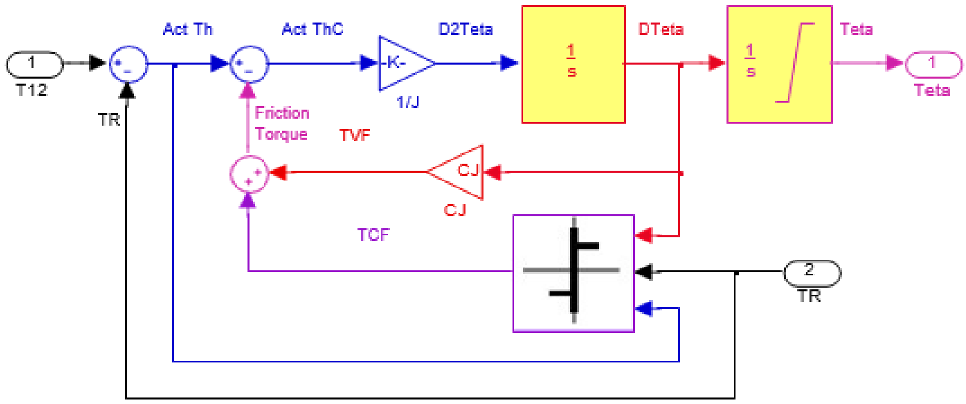

Fig. 6. MATLAB-Simulink block diagram of the flap actuation

3 Results

The numerical model illustrated in figure 6, containing the aforementioned friction calculation routine, allows to simulate the dynamic behavior of a flap control system equipped with final reversible or irreversible actuators and, therefore, allows to make comparisons between the different constructive solutions in the different operating conditions (modifying the aerodynamic loads, the actuation commands and the supply pressure it is, in fact, possible to simulate various situations, including specific failures of the system). In order to verify its validity, was then set up a numerical simulation program of the dynamic performance of a flap actuation system equipped with the aforementioned model.

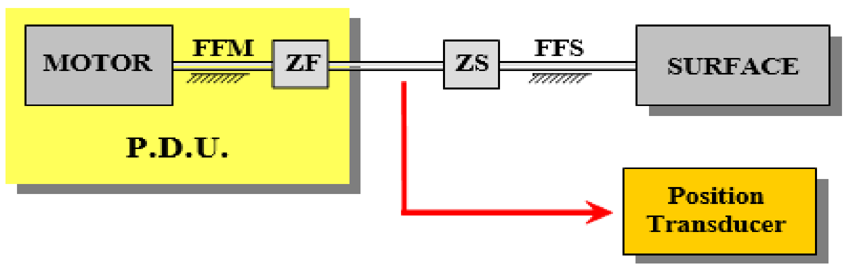

Fig. 7. Schematic representation of the passive subsystem of the flap control system actuation

The system, which obviously represents a simplification of the real one, consists of a power and control unit (PDU) which, via torsion bars, transmits power to the final actuators which drive the flaps. Each final actuator is composed of a low speed gearbox (ZS), integrated in a ball screw (BS); at the end of the torsion bar transmission are positioned the position transducers (PT) and any speed transducers (however not present in the configuration under examination). The system is controlled by an electronic control unit (ECU), not shown, on which the position control loop closes. The PDU contains the hydraulic motor, high speed gearbox (ZF), control valves. In the model examined, the presence of a tachometric dynamo was also hypothesized in the PDU, to control the actuation speed. Figure 7 illustrates the physical model of the mechanical subsystem adopted to simulate the dynamics of the flap actuation group; in its construction, particular attention was paid to the mechanical and hydraulic characteristics of the main components of the system and to the corresponding friction phenomena [21], [5], [12]. In particular, it takes into account:

• Dry friction with Coulomb model and possible clearance on the PDU (FFM), on the final actuators of the mobile surfaces (FFS) or on the position transducers (FFPT);

• Dynamic model of the servo valve, third order, with limitations on the position and on the speeds; fluid dynamic model of the control valve for hydraulic drives;

• Dynamic and fluid-dynamic model of the hydraulic motor and high-speed gearbox, which takes into account not only Coulomb friction but also viscous friction and the flow rate lost through leakage;

With the help of the corresponding numerical model, some simulations were carried out to test the validity of the proposed friction model, in particular as regards the modeling of irreversible and reversible mechanical systems subject to loads (aerodynamic loads acting on the high lift surface).

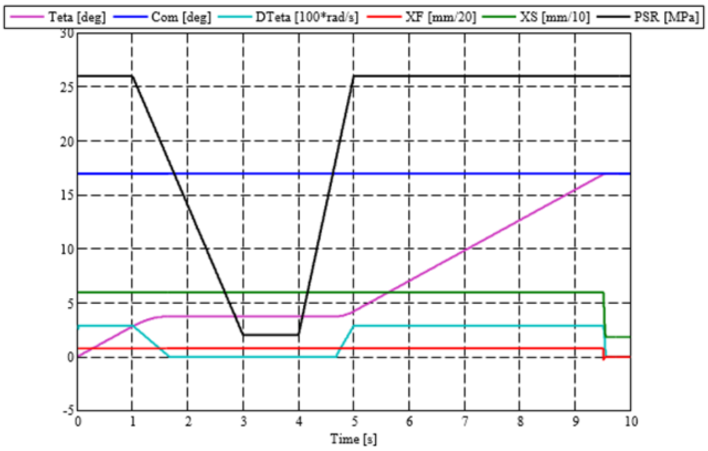

Fig. 8. Are presented the results with the use of irreversible actuator

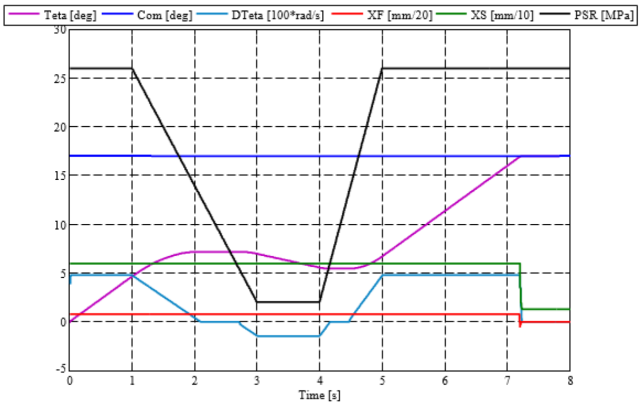

Fig. 9. In the Figure is presented, under the same operating conditions as Fig. 8, the results produced with reversible actuator.

In the figures 8 and 9, are presented two different cases of a system equipped in the case of Fig. 8 with irreversible actuator and in the case of Fig. 9 with reversible one. In both cases the Command at Time = 0 start at Com = 18 degrees.
In case of the Fig. 8 the flap responds to the required command by the rotation of the actuation shaft Teta in degree, and as it is expected in this case having an irreversible actuation the response come without oxilation.
In contrary in the case of Fig. 9 it is visible an oxilation between Time =1.5 to approximately 4.5 seconds than the response with a higher inclination angle in confront of the case in Fig. 8 reaches the required command.
In both cases, it is what is expected from the response of the system and the calculation routine of the friction involved.

The performances of the numerical simulation model in figure 5 are compared with those provided by the discontinuous Coulomb friction model (present in the Simulink libraries) and by a hyperviscous friction model; the comparison is made by integrating these friction models into the dynamic simulation program of the performance of an electro-hydraulic servomechanism and by comparing, under the same operating conditions, the temporal trends of the responses produced.

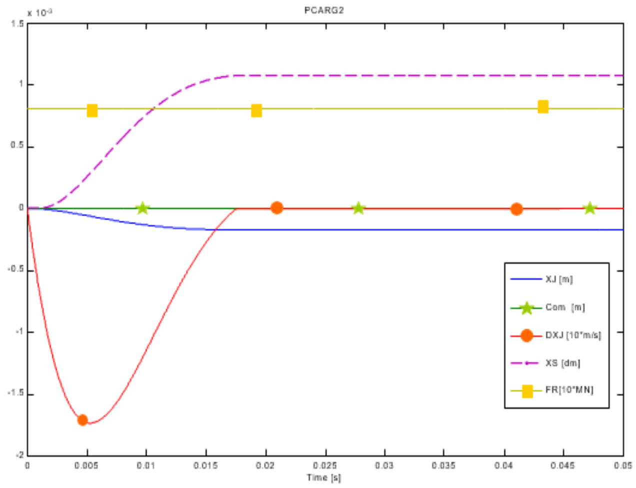

Fig. 10. Example of Coulomb friction model in relation to the velocity and to the applied active force on an electro-hydraulic servomechanism

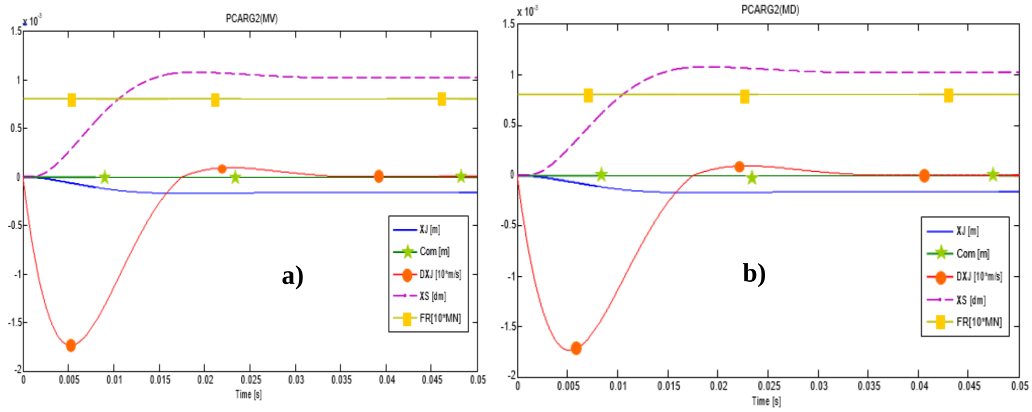

Fig. 11. Example of hyper viscous friction model a) and discontinuous Coulomb friction model b) acting on an electro-hydraulic servomechanism

In figure 10 it is represented the case of step load FR with zero initial and final value of 8000 N and constant zero command. Flow gain GQS=0.2 m^2/s. purely proportional control logic. Absence of hysteresis and leakage. Constant supply pressure. At the instant Time=0 the load of 8000 N is applied which induces an uncommanded movement of the controlled surface.

Through a dynamic behavior the system correctly moves towards a condition of equilibrium (new equilibrium at zero speed) characterized by a persistent position error obtained from the superimposition of the effects produced by the presence of the constant load FR as well as the static friction force FF (which, according to the modeling adopted, depends on the stopping conditions of the system).

It should be noted that the position error in stationary conditions would be different from that resulting from the simulation if the friction were zero; from this it can be deduced that the model proposed for the calculation of the consequences of the Coulomb friction works correctly.
In figure 11, are examined two cases, with two alternative friction models for comparisons. A "hyperviscous" friction model with saturation Figure 11 a) and a “discontinuous” friction model Figure 11 b).
There are no noticeable differences in the dynamic behavior until the overshoot peak is reached, approximately corresponding to the instant Time = 0.017 s; after this instant, both models in question are not capable of reproducing the correct condition of rest (DXJ=0) of the controlled unit obtained in Figure 10 but, erroneously, they estimate a slow convergence towards the commanded position with false progressive cancellation of the position error. The slow convergence can be explained through a sequence of speed reversals at each calculation step (numerical instability), as it certainly is in the case of Figure 11 b) and, if a viscous constant is too high, even in the case of Figure 11 a). The inability of the last two models considered ("discontinuous" and "hyperviscous") to correctly simulate the stationary condition with persistently constant position error from the moment of overshooting is evident.

4 Conclusions

- The Coulombian friction model presented in this work shows its good possibility to be used in computer programs in the simulation of servomechanisms, in which a good compromise between modeling accuracy and execution speed is required, with a consequent compactness of the algorithm.
- Both models presented in paragraph 2.1 and 2.2 present a high degree of afidability, and are easy integrated in different mechanical actuators to be used, especially aerospace one, as per high requirement associated with this mechanisms.
- All this proves the efficiency of the proposed Columbian friction Simulink models, as well as its superiority compared to other programs, including the "default" ones that are in the digital library of the Simulink program itself, greatly increases the reliability of the numerical model of the simulated servomechanism.

Used Symbols

XS	Position second stage valve	[mm/10] or [dm]
XF	Position first stage valve	[mm/10]
Com	Command	[dm] or [degree]
PSR	Differential pressure (given by the system (supply-return)) controlled pressure	[MPA]
Teta	Position	[degree]
Dteta	Radial velocity	[100*rad/s]

DXJ	Velocity	[10*m/s]
P12	Differential pressure	[10*GPa]
XJ	Response of the system	[dm]
FR	Load	[10*MN]

* Other symbols and abbreviations, are explained in the paper.

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